APPLICATIONS NOTE

MEASUREMENT OF SHAFT VIBRATION ON HYDROELECTRIC TURBINE GENERATORS

Part 1: Dynamic Behavior

BENTLY NEVADA

P.O.BOX 157
MINDEN, NEVADA USA 89423
(702) 782-3611
TELEX: 354437

PREFACE

This publication is a guide for the application of machinery protection systems on vertical rotor Francis or Kaplan (propeller) hydroelectric turbines, as well as pump storage turbines. This publication does not apply to hydroelectric turbines that use rolling element bearings.

This is Part 1 of a two-part publication. Part 1 deals with the dynamic behavior of these machines, while Part 2 deals with instrumentation applications.

INTRODUCTION

All hydroelectric turbines can be classified as being either a reaction turbine or an impulse turbine.

In an impulse turbine, all the fluid's available flow energy is converted into kinetic energy by a nozzle before the fluid strikes a moving blade(s). This occurs at atmospheric pressure. The most common impulse turbine is the Pelton wheel. Pelton wheels are typically used for high-head applications, usually ranging from 200 meters to more than 1 km.

The most common hydroelectric turbines in use throughout the world today are of the reaction type. There are two types of reaction turbines in common use -- the Francis turbine, Figure 1, and the Kaplan (or propeller) turbine, Figure 2. Streeter and Wylie [1] give a brief but accurate description of these machines:

> ...In both, all the passages flow full, and energy is converted to useful work entirely by changing the moment of momentum of the fluid. The flow passes first through the wicket gates which impart a tangential and radially inward velocity to the fluid...
>
> ...In the Francis turbine, the fluid enters the runner so that the relative velocity is tangent to the leading edge of the vanes. The radial component is gradually changed to an axial component, and the tangential component is reduced as the fluid traverses the vanes, so that at the runner exit the flow is axial with very little whirl (tangential component) remaining. The pressure has been reduced to less than atmospheric, and most of the remaining kinetic energy is reconverted to flow energy by the time it discharges from the draft tube. The Francis turbine is best suited to medium-head installations for 80 to 600 feet (25 to 180 meters)...
>
> ...In the propeller turbine, after passing through the wicket gates, the flow moves as a free vortex and has its radial component changed to axial component by guidance from the fixed housing. The moment of momentum is constant, and the tangential component of velocity is increased through the reduction in radius. The blades are few in number, relatively flat, and with very little curvature; they are so placed that the relative flow entering the runner is tangential to the leading edge of the blade. The relative velocity is high, as with the Pelton wheel, and changes slightly in traversing the blade. Propeller turbines are made with blades that pivot around the hub, thus permitting the blade angle to be adjusted for different gate openings and for changes in head. They are particularly suited for low-head installations, up to 30 meters...

Because reaction turbines are the most widely encountered, this publication will focus on the dynamic behavior of these machines.

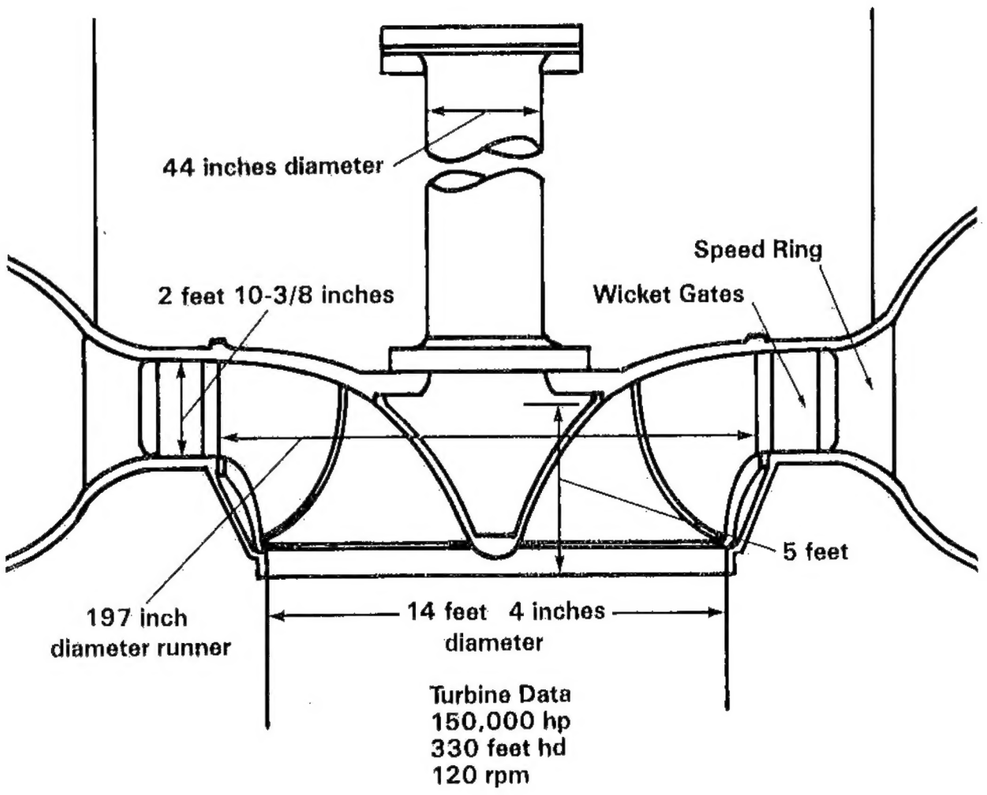

Figure 1 Francis turbine for Grand Coulee, Columbia Basin Project. (Newport News Shipbuilding and Dry Dock Co.)

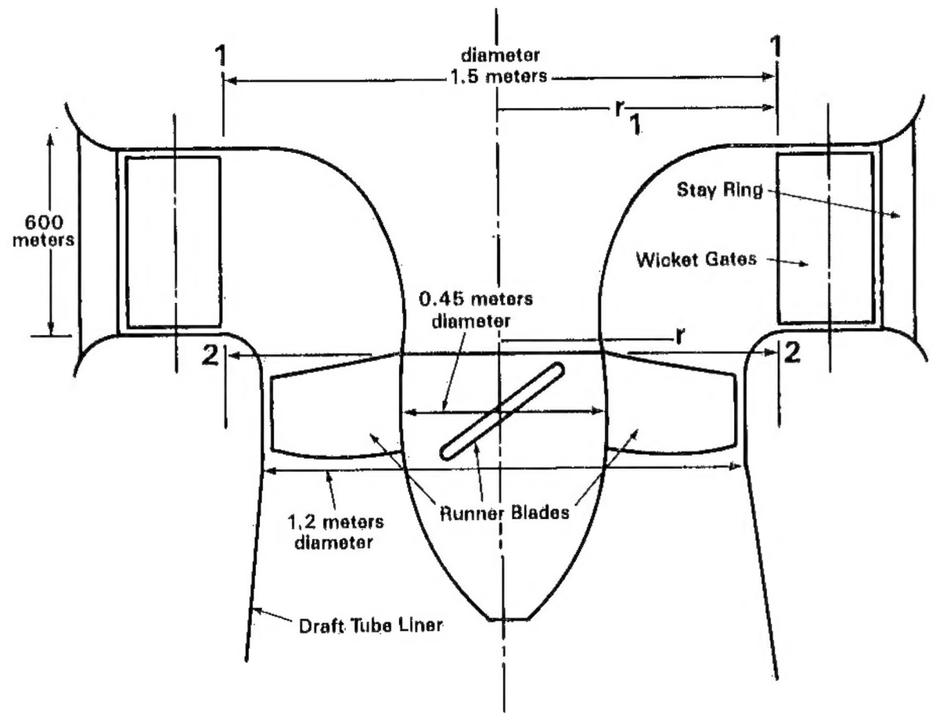

Figure 2 Schematic view of propeller turbine

DYNAMIC BEHAVIOR

Fundamentals of rotor dynamics and machinery protection systems for various types of rotating machinery are presented in other applications notes. However, vertical rotor hydroelectric turbines present some unique vibration characteristics which are not found in typical horizontal rotor or thermal machinery.

Most hydro turbines are characterized by their low running speed. Typical running speeds range from 100 to 300 rpm. In addition, due to hydraulic interaction, shaft vibration can occur over a wide frequency range. There are three major sources of shaft vibration in hydro turbines:

1. **Vibration as a Result of Mechanical Perturbation Forces**
   Mechanical perturbation forces include forces due to mass unbalance in the rotating system, elastic forces within the rotor, and frictional forces as a result of interference of rotating parts.

2. **Vibration as a Result of Hydraulic Perturbation Forces**
   Hydraulic perturbation forces include forces caused by hydraulic imbalance, hydrodynamic effects in bearings and runner seals, vortex formation and shed vortices, draft tube surge, nonuniform velocity profiles, and cavitation and flow separation in the runner.

3. **Vibration as a Result of Electrical Perturbation Forces**
   Electrical perturbation forces include forces caused by a

rotating magnetic force vector, unequal generator air gap, short-circuiting in the pole windings and/or generator bus bars, and asynchronization.

Many hydroelectric turbines are used for peak load demand, frequency, and power control. Transient operation is frequency encountered. VDI [2] states, "Larger shaft motion during the transient period can occur at start-up, at each loading or unloading, and at each fill-up or emptying of the hydraulic machine. Such events can, at peak load or pump-storage equipment, become so frequent that the sum of the time intervals of increased shaft motion amounts to more than one percent of the overall operating time. These frequent transient operating conditions must then be evaluated separately with respect to their effect on the bearings and other parts of the machine."

STEADY-STATE VIBRATION CHARACTERISTICS

Shaft vibrations during steady-state operation are often periodic in nature. Consequently, each of the various perturbation forces can be identified by its frequency response. Once defined, these frequencies can aid in the selection of the vibration monitoring system.

VIBRATION FREQUENCIES ASSOCIATED WITH MECHANICAL PERTURBATION FORCES

Forces resulting from mass unbalance manifest their presence in 1X frequency response. Figure 3 shows a cascade plot obtained during rundown of a typical Francis turbine exhibiting mass unbalance. Note that the maximum amplitude is dependent on the rotational speed.

Elastic forces occur as a result of misalignment or distortion of the shaft centerline. This can be attributed to angular misalignment between the turbine and generator shafts, parallel misalignment between the turbine and generator shafts, inclination of the shaft axis, inclination of the thrust runner, nonuniform thrust runner or thrust bearing, clearance within the shaft couplings and nonparallel journal and guide bearing surfaces.

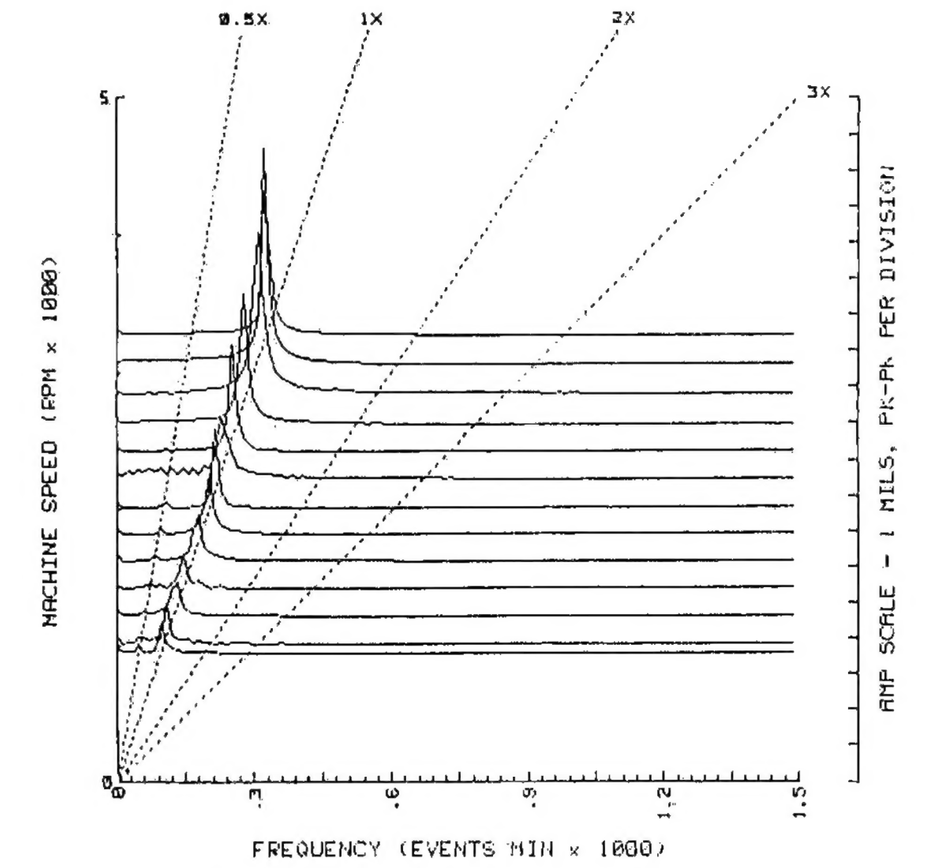
PLANT ID: WATER
TRAIN ID: MARY LAKE
MACHINE ID: HYDRO TURBINE
RUNDOWN PROBE ID: BYD RELATIVE PROBE
RUN: 1 DATE: 7 MAY 81 TIME: 1143
0.5X
1X
2X
3X
MACHINE SPEED (RPM x 1000)
AMP SCALE - 1 MILS, PK-PK PER DIVISION
FREQUENCY (EVENTS MIN x 1000)
COAST DOWN DATA FROM 327 RPM TO 100 RPM, 7.5 MEGAWATTS.

Figure 3

Incorrect centering and angularity between the turbine and generator shafts will impede the free rotation of the shaft. Perturbation forces, due to shaft elasticity, are transmitted to the guide bearings. The fundamental frequency of vibration may be 1X, but is predominately 2X. The behavior of the maximum amplitude can be constant or vary with time. According to VDI 2059 [2], the maximum amplitude can change with speed and load. Vladislavlev [3] states that when multipad thrust bearings are used, the fundamental frequency would be kn/60 in Hz, where k is the number of thrust pads, and n is the running speed in rpm.

Figure 4 shows a plot of amplitude versus order of rotational speed for the lower guide bearing of a Francis turbine.

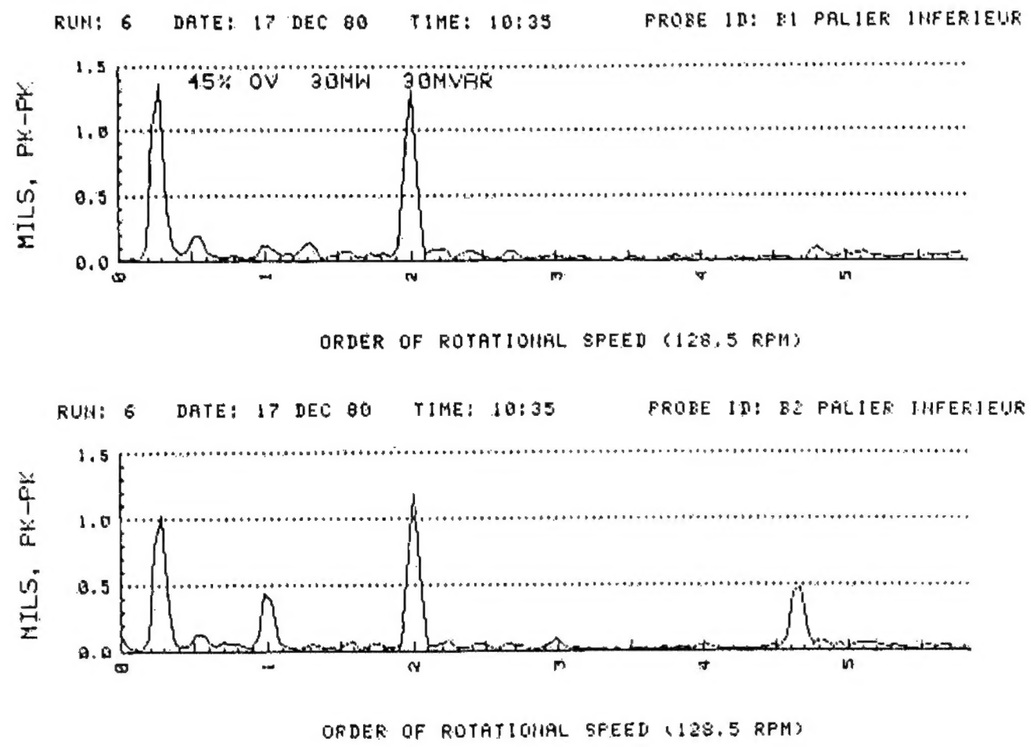
RUN: 6 DATE: 17 DEC 80 TIME: 10:35 PROBE ID: B1 PALIER INFERIEUR
45% OV 30MW 30MVAR
MILS, PK-PK
ORDER OF ROTATIONAL SPEED (128.5 RPM)
RUN: 6 DATE: 17 DEC 80 TIME: 10:35 PROBE ID: B2 PALIER INFERIEUR
MILS, PK-PK
ORDER OF ROTATIONAL SPEED (128.5 RPM)

Figure 4

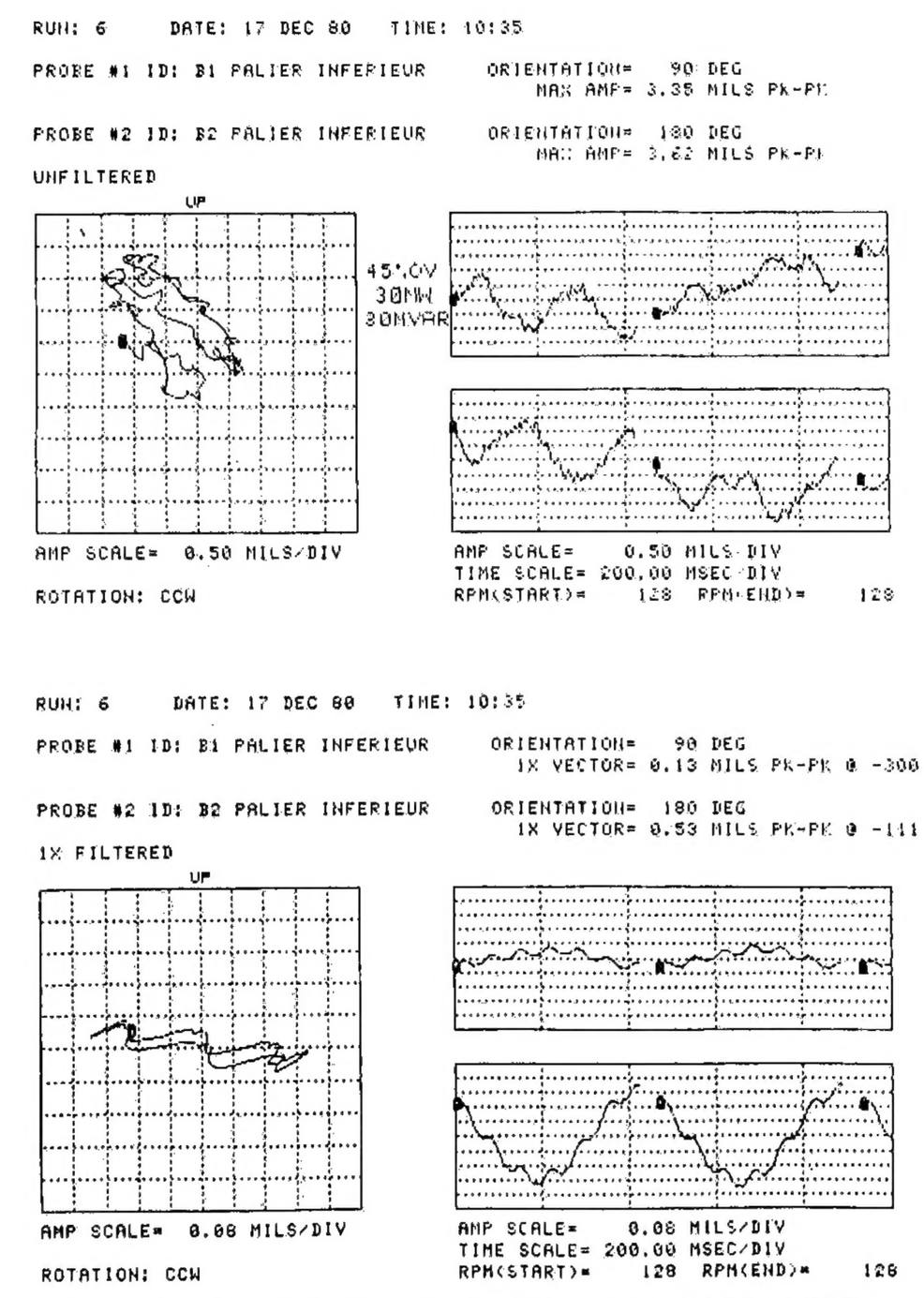
RUN: 6 DATE: 17 DEC 80 TIME: 10:35
PROBE #1 ID: B1 PALIER INFERIEUR ORIENTATION= 90 DEG
MAX AMP= 3.35 MILS PK-PK
PROBE #2 ID: B2 PALIER INFERIEUR ORIENTATION= 180 DEG
MAX AMP= 3.82 MILS PK-PK
UNFILTERED
UP
45°OV
30MW
30MVAR
AMP SCALE= 0.50 MILS/DIV
ROTATION: CCW
AMP SCALE= 0.50 MILS/DIV
TIME SCALE= 200.00 MSEC/DIV
RPM(START)= 128 RPM(END)= 128
RUN: 6 DATE: 17 DEC 80 TIME: 10:35
PROBE #1 ID: B1 PALIER INFERIEUR ORIENTATION= 90 DEG
1X VECTOR= 0.13 MILS PK-PK @ -300
PROBE #2 ID: B2 PALIER INFERIEUR ORIENTATION= 180 DEG
1X VECTOR= 0.53 MILS PK-PK @ -111
1X FILTERED
UP
AMP SCALE= 0.08 MILS/DIV
ROTATION: CCW
AMP SCALE= 0.08 MILS/DIV
TIME SCALE= 200.00 MSEC/DIV
RPM(START)= 128 RPM(END)= 128
NOTE: BOTH ORBITS COMPENSATED FOR NON-VERT/HORIZ PROBE ORIENTATION

Figure 5

According to the Mechanical Engineering Services Report [4], "...the steady-state vibration frequency characteristics of the generator are nearly identical under all load conditions. A strong 2X rpm vibration signal suggests a preload condition. This is further confirmed by the restricted orbit noted at the pallier interieur (lower generator guide bearing) probe location." Figure 5 shows the corresponding orbit and time-based vibration signal.

Shaft vibrations can occur due to dry friction. This is also known as a rub. A rub can result from inadequate lubrication or excessive bearing clearances. Contact can be initiated by increased mass unbalance, thermal bow or other mechanisms causing the shaft to deflect or bend beyond the available clearances. This can also be caused by misalignment of stationary elements.

Vladislavlev [3] gives a detailed explanation of a full rub in a hydro turbine (Figure 6). He says, "During rotation, shaft B is in contact with bearing A. Frictional force K′ acts on the bearing and the force K′ acts on the shaft at the point of contact. If K′ is replaced by a parallel force acting at the journal center with the corresponding moment, K′r where r is the journal radius. The moment K′r, causes torque on the shaft, and the force K′ applied at the center deflects the shaft." This causes the shaft to rotate in a direction counter to the rotation of the shaft. This results in extreme vibration. Vladislavlev goes on to state that "the frequency associated with this condition varies from 2X to 5X for hydro turbines."

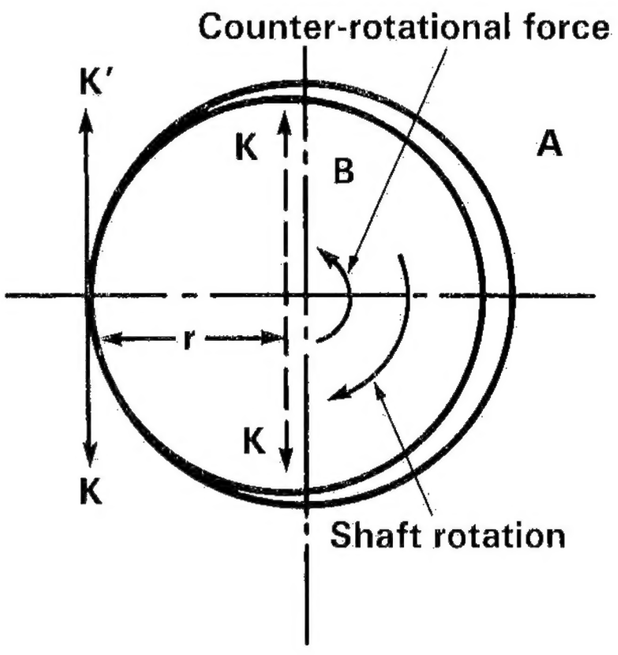

Figure 6 Frictional forces on a rotating shaft

VIBRATION ASSOCIATED WITH HYDRAULIC PERTURBATION FORCES

Hydraulic interaction between the moving fluid and turbine components will cause vibration not only on the components themselves but within the entire rotating system. During steady-state operation, certain vibration frequencies can be associated with specific hydraulic perturbations.

VORTEX INTERACTION

According to Vladislavlev [3], "Poor construction of the inlet portion of the spiral casing (presence of dividing piers close to the stator ring, stay vanes, etc.) is conducive to the formation of vortices under certain operating conditions. The individual vortices combine together to form intertwining vortex filaments. On reaching the runner blades they are broken up and create perturbation forces. The intensity of the respective forces, which depends on the size of the vortex filament, can be quite large. In such a case, the first harmonic frequency of the perturbation force corresponds to f = nz/60, where z is the number of blades and n is the rotating speed of the hydro unit shaft. Basically, this type of action of the flow on the runner is met within low-head power plants having adjustable-blade hydraulic turbines."

In Francis turbines where the runner is very close to the wicket gate exit, the runner is subjected to hydraulic perturbations. This is caused by boundary layer separation and the formation of vortex filaments after the wicket gates. The fundamental frequency of the corresponding vibration is nzo/60, where zo is the number of wicket gates. According to another Mechanical Services Engineering Report [5], "the highest amplitude vibration encountered at the turbine bearing was at near full-load operation. Also, high frequency vibration is indicated at the same load condition at a 20 times rotational frequency... There are 20 wicket gates on the turbine... The amplitude of the 20X component is highest when the 1X component is highest." Figure 7 shows amplitude versus order of rotational speed and the vibration associated with this phenomena.

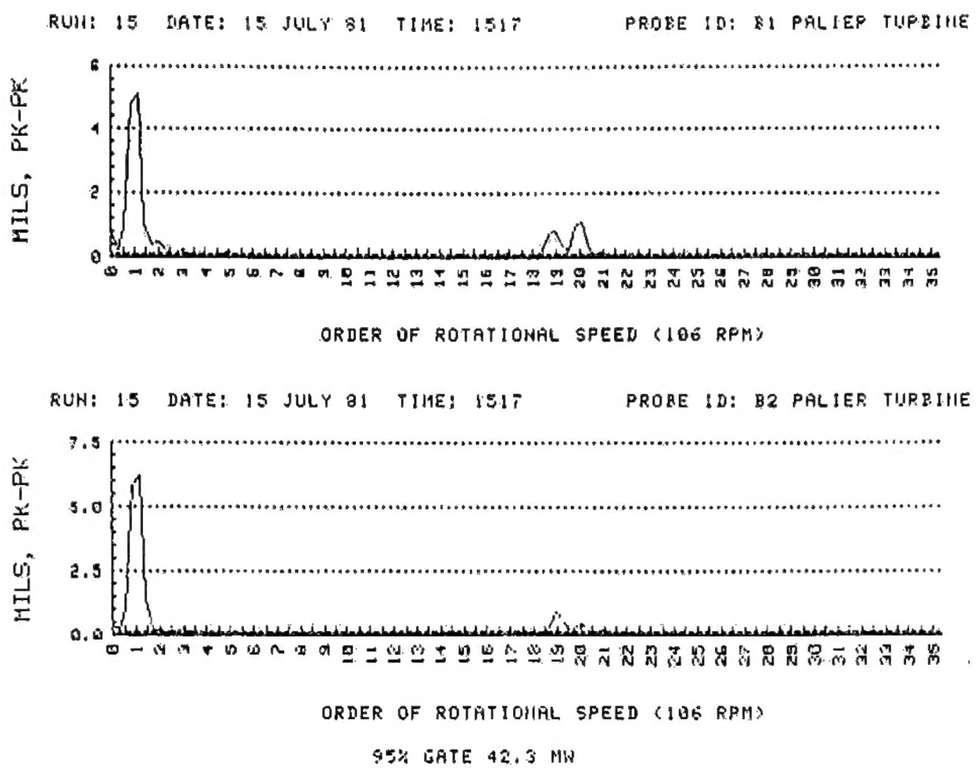
RUN: 15 DATE: 15 JULY 81 TIME: 1517 PROBE ID: B1 PALIER TURBINE
MILS, PK-PK
ORDER OF ROTATIONAL SPEED (106 RPM)
RUN: 15 DATE: 15 JULY 81 TIME: 1517 PROBE ID: B2 PALIER TURBINE
MILS, PK-PK
7.5
5.0
2.5
0.0
ORDER OF ROTATIONAL SPEED (106 RPM)
95% GATE 42.3 MW

Figure 7

A very common occurrence associated with Kaplan and Francis turbines is known as the "Rough Load Zone" or "Rheingans Influence." Vladislavlev [3] states that research on turbines under normal operating conditions shows that spiral vortex filaments form after the runners themselves rotate. The direction of rotation depends both on the velocity of the flow and on the angular velocity of the vortex. Vladislavlev states:

> The spiral vortex filaments have different rotational speeds with respect to the vertical axis. This rotational speed depends on the operating regime of the turbine and usually differs from the speed of the runner, being somewhat lower. Figure 8, which shows the results of tests on a turbine with a 4-meter diameter runner under actual operating conditions, shows that the rotational speed of the spiral vortex filament is lower than that of the runner.
>
> The rotation spiral vortex filaments downstream of the runner alter the velocity and pressure distribution. This gives rise to pressure pulsations in the flow passage of the turbine, subjecting the runner blades to varying pressure forces.
>
> Tests show that these vortex filaments appear when the load on the turbine is within 25 to 45 percent of the nominal. Under these operating regimes, the vibration of the turbine increases somewhat....
>
> ....According to Donaldson's findings [6], for draft tubes of normal depth, the frequency of pressure pulsations downstream of the runner due to the break-up of the vortex filament may be approximated by the formula:

$$f = \frac{\frac{n}{60\,k}}{} = \frac{n}{60 \times 3.6\ \text{Hz}}$$

It should be noted that:

1. The frequency of pressure variations in the draft tube depends not only on the speed of the turbine shaft, but also on the load on the turbine.

   (For a Francis turbine, it may be assumed for preliminary calculations that the coefficient K in Donaldson's formula equals 3.6 and for adjustable-blade turbines K = 4.2 to 4.6.)

2. Intensity of pressure in the draft tube depends on the level of water in the tailrace.

Figures 9 and 10 are the results of data obtained by Mechanical Engineering Services [4] showing the effects of Rheingans Influence. The data was obtained at the upper guide bearing of a Francis turbine operating at 40 percent gate position. Notice the large amplitude occurring at approximately 0.25 X.

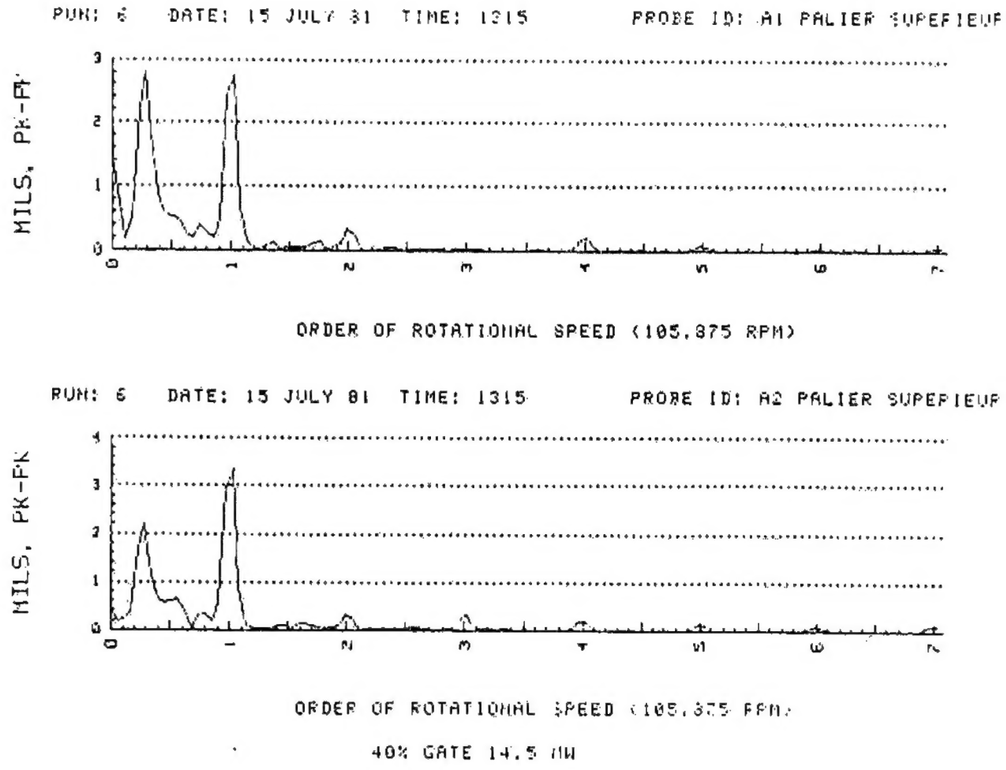
RUN: 6 DATE: 15 JULY 81 TIME: 1315 PROBE ID: A1 PALIER SUPERIEUR
MILS, PK-PK
ORDER OF ROTATIONAL SPEED (105.875 RPM)
RUN: 6 DATE: 15 JULY 81 TIME: 1315 PROBE ID: A2 PALIER SUPERIEUR
MILS, PK-PK
ORDER OF ROTATIONAL SPEED (105.875 RPM)
40% GATE 14.5 MW

Figure 9

CAVITATION

As water moves through the runner, it flows into an area where its pressure is reduced to vapor pressure, resulting in the formation of a vapor wicket. As the flow progresses through the runner, the vapor pocket travels with the flow. Upon reaching an area of higher pressure, implosion occurs. If the vapor bubbles are in contact with the runner when they

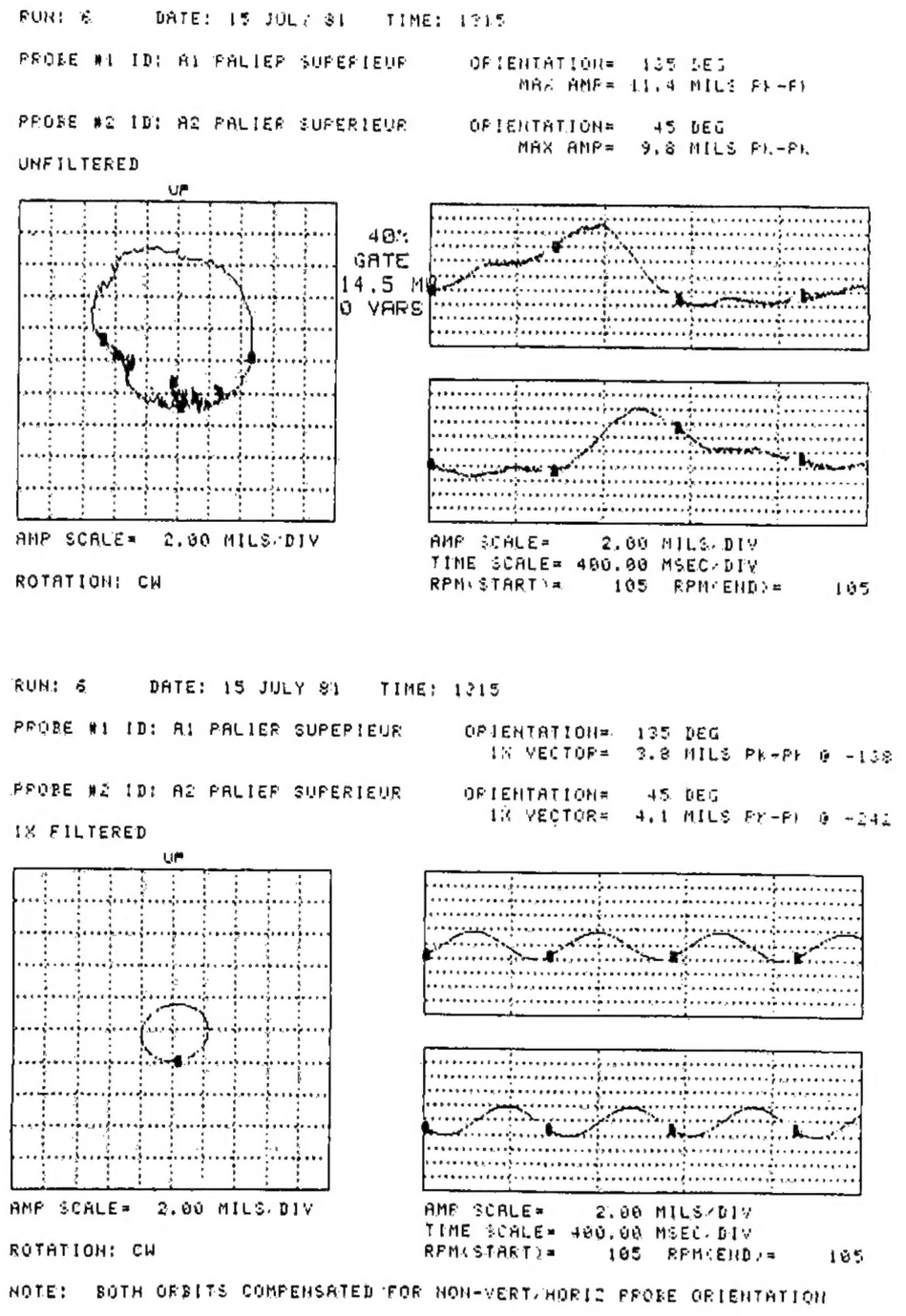
RUN: 6 DATE: 15 JULY 81 TIME: 1315
PROBE #1 ID: A1 PALIER SUPERIEUR ORIENTATION= 135 DEG
MAX AMP= 11.4 MILS PK-PK
PROBE #2 ID: A2 PALIER SUPERIEUR ORIENTATION= 45 DEG
MAX AMP= 9.8 MILS PK-PK
UNFILTERED
UP
40% GATE 14.5 MW 0 VARS
AMP SCALE= 2.00 MILS/DIV
AMP SCALE= 2.00 MILS/DIV
TIME SCALE= 400.00 MSEC/DIV
ROTATION: CW
RPM(START)= 105 RPM(END)= 105
RUN: 6 DATE: 15 JULY 81 TIME: 1315
PROBE #1 ID: A1 PALIER SUPERIEUR ORIENTATION= 135 DEG
1X VECTOR= 3.8 MILS PK-PK @ -138
PROBE #2 ID: A2 PALIER SUPERIEUR ORIENTATION= 45 DEG
1X VECTOR= 4.1 MILS PK-PK @ -242
1X FILTERED
UP
AMP SCALE= 2.00 MILS/DIV
AMP SCALE= 2.00 MILS/DIV
TIME SCALE= 400.00 MSEC/DIV
ROTATION: CW
RPM(START)= 105 RPM(END)= 105
NOTE: BOTH ORBITS COMPENSATED FOR NON-VERT/HORIZ PROBE ORIENTATION

Figure 10

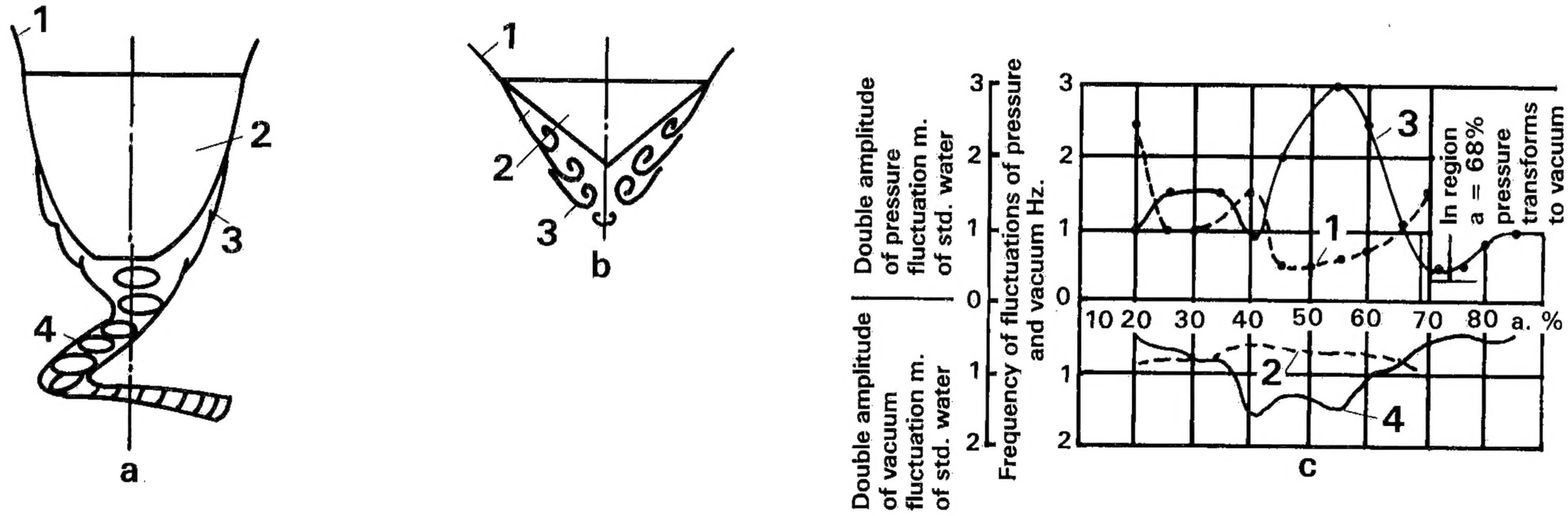

Figure 8 Influence of shape of body in flow on vibrations of turbine:
(a and b) convex linear profiles of runner hub: 1-runner; 2-runner hub; 3-zone of flow separation and vortex formation; 4-vortex filaments; c-frequency and amplitude of variations in pressure in draft tube of turbine of 34,000 KW operating at head of 32.5 m (rotational frequency 1.78 Hz): 1-frequency of pressure variations at periphery of tube; 2-frequency of pressure variations at center of draft tube; 3-double amplitude of pressure variations at periphery of tube; 4-double amplitude of pressure variations at center of tube.

Note: The discrete values of the frequency have been arbitrarily connected by the broken line.

collapse, the forces exerted by the implosion create very high localized pressures that cause pitting and fatigue of the runner. Cavitation will also reduce the operating efficiency of the turbine. According to one report [7], the frequency of vibration caused by cavitation could be as high as 100 KHz.

HYDRAULIC UNBALANCE

According to Vladislavlev [3], violation of symmetrical pressure distribution at the lateral surface of a radial-axial turbine runner gives rise to a variable perturbation force on it, known as hydraulic imbalance. Hydraulic imbalance of a runner occurs when:

1. The gland seal rings on the runner wear out eccentrically.
2. The rim of the runner is machined in order to remove metal from one side during balancing by manufacturers.
3. The shape of the balancing mass attached to the outer surface of the rim of the runner is wrong, so it disturbs the symmetry of pressure distribution on the lateral surface of the runner.

A Mechanical Engineering Services Report [5] identifies hydraulic unbalance as occurring at a frequency of 1X.

HYDRAULICALLY-INDUCED AXIAL VIBRATION

In the special case of Francis turbines where the thrust bearing is located above the generator, vertical misalignment between the runner and wicket gates can result in axial vibration. This misalignment can be caused by axial rotor expansion caused by thermal growth in the generator area. Vladislavlev [3] explains the cause and effects:

> Asymmetrical positioning of the runner passages in relation to the height of the wicket gates is conducive to separation of flow from the inner surface of the rim of the runner. This leads to pressure pulsations in the runner and vibration of the turbine. The separation of flow is considerably facilitated by large curvature of the surface at the lower rim and positioning the runner slightly lower in relation to the wicket gates. (Figure 11). Separation begins at a certain load and with increasing wicket gate openings (increased loads), the zone of separated flow becomes larger and turbine vibrations increase.

Asymmetrical positioning of the runner with respect to the wicket gates in the vertical direction leads to redistribution of pressure ahead of the seal gaps. The pressure before the gaps increases where the gland seal protrudes into the flow (Figure 11). This increases the leakage through the gland seals, and they wear out more rapidly than when the sealing is located below the edge of the wicket gate opening. Due to separation of flow before the seal, the pressure decreases and the leakage through the seal is reduced. The nonuniform wear of the gland seal gap leads to changes in the axial hydraulic force exerted on the runner. Unequal wear of the seals may lead to a situation where due to rapid wear (of the lower seal, for instance), the axial component of hydraulic force is directed upward. Under certain load conditions, this may lift the rotor of the hydro unit. This phenomenon was observed in the turbines of two high head power plants.

The values of gland seal wear observed on turbine runners of 1460 mm (9500 kw, 200 m head) after 2,000 hours operation were:

> 0.9 mm at the upper ring and 2.2 to 2.6 mm at the lower ring against a designed gap of 0.75 mm.

Under these conditions, it was observed that the rotor was lifted in the load range of 0 to 6,000 kw. The rotor lift led to the fusing of the lower portion of the turbine bearing as a result of friction at the rim of the oil scraper ring. The rotor lifted smoothly and was accompanied by some increase in the vertical vibrations.

The frequency of vertical vibrations in this case was nearly 2.5 Hz.

RUNNER BLADE VIBRATION

Donaldson [6] conducted research on the relationship between the trailing edge shape of runner blades and their effect on vibration in Francis runners. According to Donaldson, "The most common type of vibration in hydraulic turbines appears to be a runner vibration excited by the shedding of the vortices of the Von Karman trail from the trailing edges of the runner buckets."

When the frequency of the Von Karman trail vortices matches the natural frequency at the runner blade exit, resonance will occur. The natural frequency of the runner blade exit can be determined experimentally. Donaldson was able

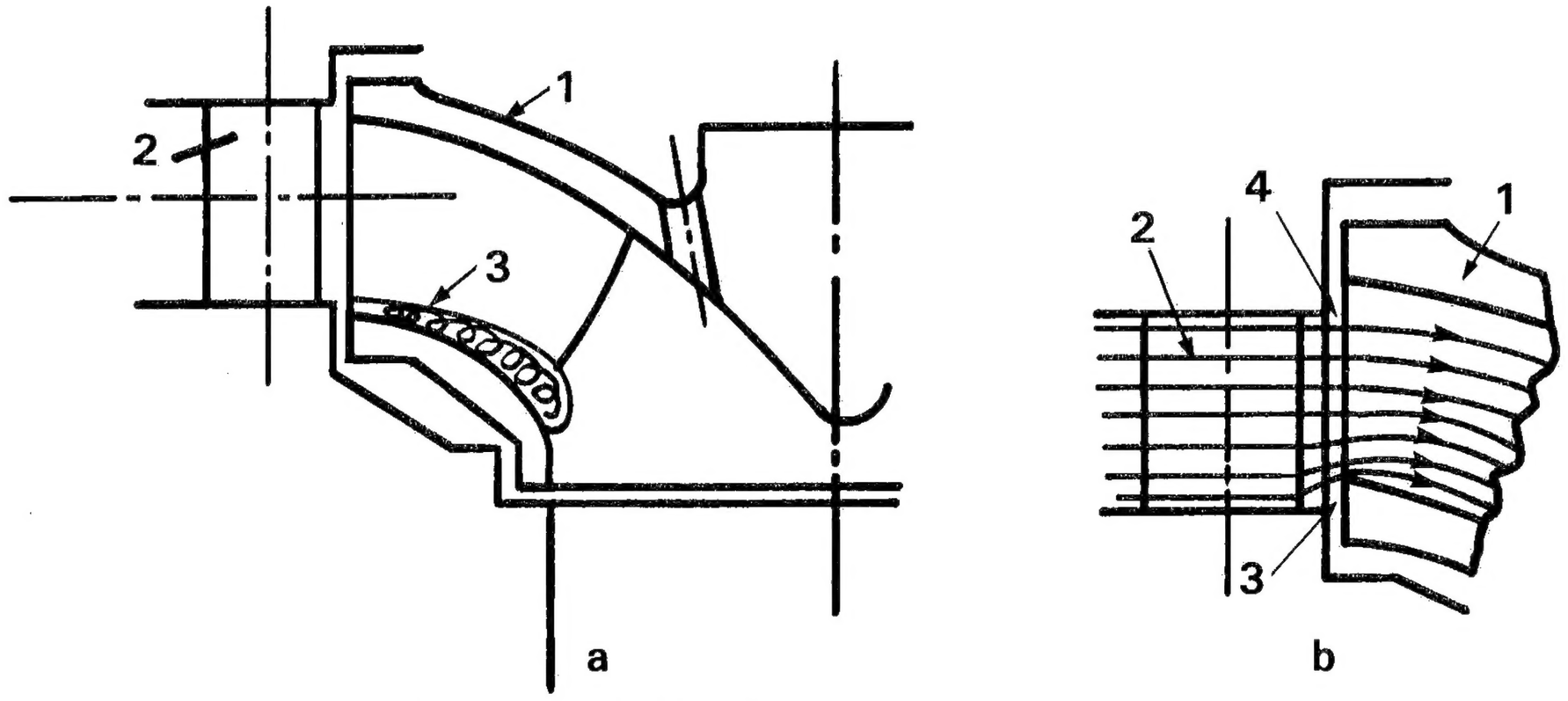

Figure 11 Diagram of flow with improper vertical location of runner:
a-separation of flow from inner surface of lower rim when runner is below normal position; 1-runner; 2-guide vanes of wicket gate; 3-zone of flow separation; b-formation of high-pressure zone before gap of lower gland seal of runner: 1-runner; 2-guide vanes of wicket gate; 3-lower seal gap; 4-upper seal gap.

to demonstrate that when wetted, the natural frequency is reduced by approximately 15 percent due to water damping. Consequently, if the natural frequency of the blade is known, it may be possible to identify blade vibration caused by the Von Karman trail. As an example, for a 23,500 hp Francis turbine with a 210 Hz natural blade frequency (dry), the vortex frequency was calculated to be 60 to 176 Hz. A large amplitude of vibration was found to be at 180 Hz.

Sagawa [7] showed that boundary flow separation on the leading vertical runner surface also caused high frequency vibrations in the runner of approximately 200 to 450 Hz.

VIBRATION ASSOCIATED WITH ELECTRICAL PERTURBATION FORCES

Vibration can be caused by magnetic unbalance. Vladislavlev [3] states:

> The current flowing through the windings of the generator rotor establishes a magnetomotive force on the rotor and magnetic induction in the air gap. Radial forces between rotor and stator appear around the periphery of the rotor. These try to bring them closer together...the variable component of magnetic force acts on the rotor at pole frequency.
>
> An asymmetrical air gap between the rotor and stator (for instance, when the stator is elliptical due to radial displacement of individual poles, etc.) leads to a unidirectional radial force directed toward the point where the gap is smaller.

The fundamental frequency of the vibration caused by variation in the perturbation forces acting on the rotor or stator are:

ni/60 or 2 ni/60 Hz

where n is the shaft speed in rpm and i is 1.2.3...

According to VDI [2], the behavior of the maximum amplitude with time is constant and the maximum amplitude is slightly power dependent. In some cases, the maximum amplitude may decrease at excitation when, for example, the vector of the residual unbalance is opposite to that of the magnetic unbalance. However, in most cases, the shaft vibration will increase with increased excitation.

Vibration can also be caused by a pole winding short circuit. When this occurs, air gap induction becomes nonuniform, which can cause vibration at 1X or its harmonics. Like vibration caused by magnetic unbalance, the vibration associated with a pole winding short circuit is dependent on excitation and the maximum amplitude is constant with time.

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Types of Forces	Perturbation Forces Forces and Cause of Vibrations	Frequency		Types of Hydroturbines
		Formula	Magnitude, Hz	
1	2	3	4	5
	Centrifugal forces due to rotor imbalance	n/60	1 to 5 1 to 12.5 5 to 12.5	PL (Kaplan) RO (Francis) K (Pelton wheel)
	Elastic forces of shaft	n/60; 2n/60	1 to 10 1 to 25 5 to 25	PL RO K
	Frictional forces	(2 to 5)n	5 to 40	for all types
	Nonuniform velocity distribution and vortex formation	$\frac{nz_{runner}}{60}$	4 to 20 10 to 80	PL RO
	Hydraulic imbalance of runner	n/60	1 to 12.5	RO
	Cavitation phenomenom in turbine	---	0.5 to 100	for all types
	Vortex formation in draft tube	n/60k k = 4.2 to 4.6 n/60k k = 3.6 to 4.2	0.2 to 1.5 0.3 to 4	PL RO
	Variable components of moments due to one nozzle in Pelton wheel	$\frac{nz_{runner}}{60}$	60 to 300	K
	Periodic component of magnetic drag	$2f$	100	for all types
	Variation in air gap	n/60; 2n/60	1 to 25	for all types
	Short circuiting of rotor pole windings	n/60	1 to 12.5	for all types
	Asymmetrical loading of generator	$2f$	100 (200-300)	for all types
1	2	3	4	5
	Asynchronous operation without excitation	$2f_s$	90	for all types
	Asynchronous operation with excitation	f_s ; $2f_s$	45;90	for all types
	Continuous short circuit of one phase of generator	$2f_e$	100	for all types
	Continuous short circuit of two phases of generator	$2f_e$	100	for all types

Figure 12 Summary of Vibration Frequencies Removed from [3].

APPLICATIONS NOTE

MEASUREMENT OF SHAFT VIBRATION ON HYDROELECTRIC TURBINE GENERATORS

Part 2: Instrumentation Applications

BENTLY NEVADA

P.O.BOX 157
MINDEN, NEVADA USA 89423
(702) 782-3611
TELEX: 354437

PREFACE

This publication is a guide for the application of machinery protection systems on vertical rotor Francis or Kaplan (propeller) hydroelectric turbines, as well as pump storage turbines. This publication does not apply to hydroelectric turbines that use rolling element bearings.

This is Part 2 of a two-part publication. Part 1 deals with the dynamic behavior of these machines and should be considered a prerequisite to this publication. Part 2 deals with the selection and installation of monitoring equipment.

INTRODUCTION

The information and techniques presented in this publication are based on the combined experiences of Bently Nevada's Mechanical Engineering Services, Applications Engineering, and Sales Departments.

SHAFT RELATIVE VERSUS ABSOLUTE

For low frequency measurements associated with hydroelectric turbines monitoring, accurate amplitude and phase information is very difficult to obtain using case-mounted velocity transducers. It has been determined [1], [2], that when the casing amplitude is less than or equal to 20 percent of the shaft amplitude, casing measurements may not be required. When casing motion is negligible, proximity probes should be used.

RADIAL VIBRATION MEASUREMENT PLANES

Figure 1 shows the recommended probe locations for a typical Francis turbine for radial vibration monitoring. In most designs, either two or three guide bearings are encountered. Probes should be installed as close as possible to each bearing location in RVXY fashion as shown.

Usually, adequate runner vibration can be detected at the turbine guide bearing. However, in some cases, it may be necessary to directly monitor vibration on the runner band itself. This probe location may be warranted for machines which have a history of runner-related failures or seal clearance problems. Special underwater runner probes are available from Custom Products. (Applications should be evaluated on an individual basis by Custom Products Pre-Order Engineering Group). Figure 2 shows an accepted method of runner probe installation. The probes should be recessed in a counter bore and back-filled with epoxy to protect the probe tip from hydraulic abuse. Runner probes should also be mounted in standard RVXY fashion. However, due to high levels of runout, it is sometimes necessary to use four probes radially opposed and a summing amplifier to obtain more accurate data.

THRUST PROBE INSTALLATION

Axial thrust position is the measurement of the relative position of the thrust collar to the thrust bearing. This measurement is perhaps one of the single most important monitored

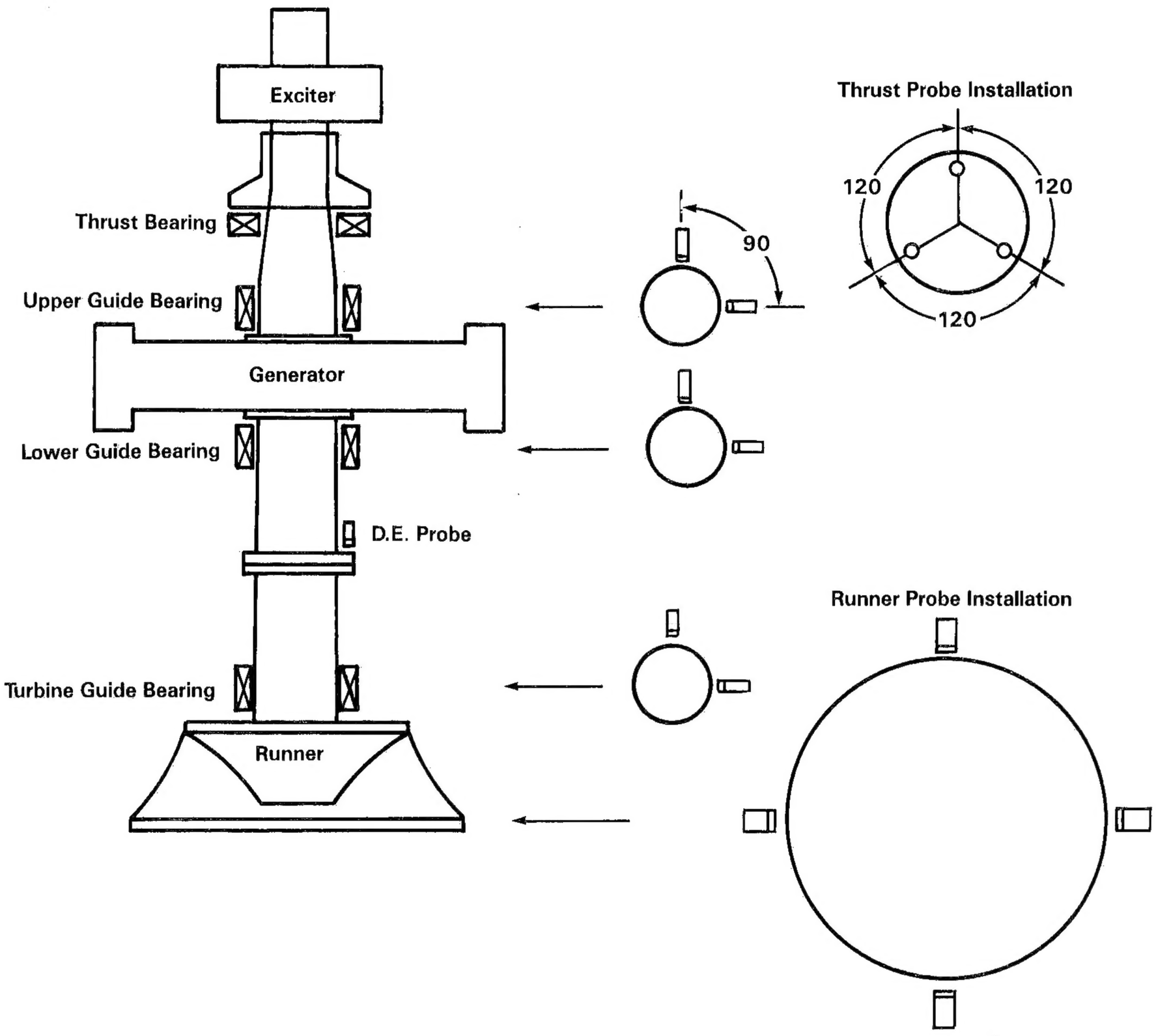

Figure 1 Francis turbine probe installation diagram

parameters on hydroelectric turbines. The primary purpose of axial position measurement is to ensure against an axial rub. Axial thrust bearing failures can be catastrophic, and every attempt should be used against this type of failure.

Three probes should be located as shown in Figure 1. Each of these probes should be mounted 120 degrees apart. Two of these probes are to be used for continuous thrust position monitoring while the third probe can be used for differential expansion. Three probes located at the thrust runner offer the added advantage of determining the thrust plane. This can be useful for diagnostic purposes, such as vertical alignment and shaft inclination.

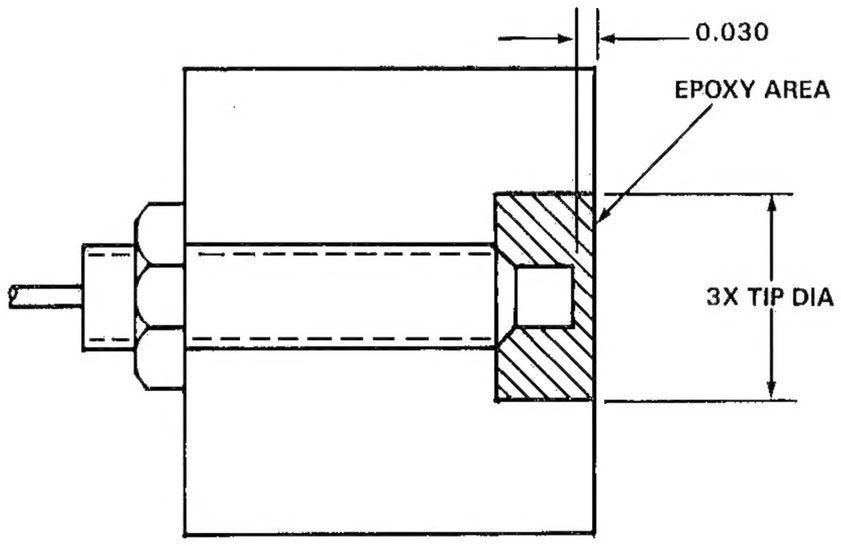

Figure 2 Runner probe installation

DIFFERENTIAL EXPANSION PROBE INSTALLATION

Francis turbines with the thrust bearing located above the generator are widely used. For these machines, a differential expansion measurement is recommended. During steady-state operation, thermal growth in the generator shaft can result in vertical runner misalignment and seal clearance reduction. This can result in accelerated seal wear. This can also lead to highly destructive vertical vibrations as mentioned in Part 1 of this publication. While thrust probes could detect the effective vibration, a differential expansion measurement could provide predictive information.

Probe installation can be accomplished as shown in Figure 1. On many hydroelectric turbines, the coupling between the turbine shaft and generator shaft is covered or shielded. This provides an excellent target for the lower probe. The upper probe can be situated on the thrust runner as shown. For a complete discussion of differential expansion, refer to Applications Note, "Differential Expansion Applications."

TEMPERATURE

In hydroelectric turbines that operate at higher than average running speeds, babbitt temperature is normally monitored using one Resistance Temperature Detector (RTD) located at both the leading and trailing edges of each guide and thrust bearing pad. Lube oil temperature is also continuously monitored. It is critical that RTDs can be located as near as possible to the babbitt to avoid problems associated with thermal transients.

KEYPHASOR

In many situations, it may be difficult to find a once-per-turn reference point on the shaft. When a keyway is not available, a steel notch can be attached to the shaft with epoxy.

MONITORING EQUIPMENT

All guide bearing and runner transducer signals should be continuously monitored using Bently Nevada 7200 Series RVXY-II monitors with eccentricity position alarm modules. While the RVXY-II monitors vibration, the eccentricity position alarm module can provide protection against excessive preload in a situation where the shaft is jammed hard against the side of a bearing and radial vibration levels are not abnormal. For hydroelectric turbines with typical operating speeds of 100 to 300 rpm, a low frequency modification is available for 30 to 36,000 rpm frequency response.

Thrust position transducer signals should be continuously monitored using Bently Nevada 7200 Series Thrust Position or Dual Voting Thrust Position monitors. Differential expansion transducer signals should be continuously monitored using a Bently Nevada 7200 Series Differential Expansion Monitor.

Since many hydroelectric turbine vibrations can be directly attributed to generator load, a Bently Nevada 7200 Series Process Variable Monitor can be used to help plant personnel correlate vibration as a function of load. Meters can be provided with KW, MW, and MVAR scales.

On many hydroelectric turbines, a flyball governor is used to provide over-speed protection. A Bently Nevada 7200 Series Tach III can be used to enhance this protection by providing the capability of continuous machine speed measurement and readout. It contains two field adjustable alarms for under-speed and over-speed monitoring with both latching and nonlatching options available.

The tachometer's minimum frequency response is 5 Hz or 300 rpm at one event per revolution. The minimum rpm indication can be decreased by increasing the number of events per revolution. The drive gear located on the governor can be used as a means of obtaining the signal. It is interesting to note that typical runaway speeds are on the order of 400 rpm.

Typically, wicket gate position is controlled by a servo/hydraulic system. This system is controlled by using a rotary potentiometer in the feedback loop. Accurate wicket gate position is critical to the proper operation of the turbine. To help ensure that accurate gate positioning is accomplished, a Bently Nevada 7200 Series Valve Position Indicator can be used.

Another important consideration when monitoring hydroelectric turbines is the ability to predict incipient rotation when the unit is stopped for service or repair. The primary function of this monitor is to indicate dead stop and breakaway of turbine rotation. This is commonly referred to as "creep detection." For this application, a Bently Nevada 7200 Series *Creep Monitor should be used. To obtain the correct zero speed period and ensure proper resolution, the governor drive gear can be used to provide the input signal. Since this monitor is used to ensure personnel safety, two transducer systems should be used independently using both monitor channels.

*Creep Monitors will be available from Custom Products by January 1984.

DIAGNOSTIC INSTRUMENTATION

The Digital Vector Filter 2 (DVF 2) is an advanced design data collection instrument for dynamic diagnosis and balancing of virtually all rotating equipment. This instrument is capable of operating as low as 30 rpm with special filtering modifications.

The DVF 2 provides two continuous channels of amplitude and phase data for rotor behavior plotting and analysis through an IEEE 488 computer interface.

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